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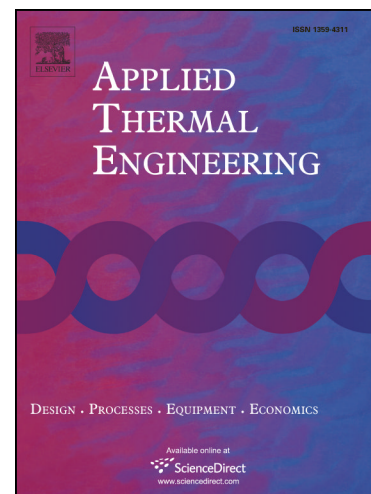
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An experimental study on condensation heat transfer characteristics of R-600a in tubes with coiled wire inserts

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Abstract

In this experimental research, condensation heat transfer enhancement of R-600a as a consequence of used spring inserts inside a horizontal condenser is studied. The condenser was a copper tube with the internal diameter, wall thickness, and length of 8.1, 0.71, and 1000 mm, respectively. Five coiled wires with various coil pitches of 10, 20 and 30 mm, and various wire diameters of 0.5, 1 and 1.5 mm were used inside the test pipe. Tests were conducted at vapor qualities within the range of 0.03-0.79 and mass velocities between 115-365 kg/m²s. Results indicated that coiled wires are advantageous instruments in heat transfer improvement of the condenser. In this regard, the coiled wire with the wire diameter of 1.5 mm and coil pitch of 10 mm showed the best performance among all inserts by the heat transfer coefficient increased up to 107% over the smooth tube. It was also observed that the use of springs affects the transition of flow pattern from annular to intermittent, and the annular regime was observable at a lower vapor quality in comparison to the smooth tube. Finally, a new correlation is developed to predict the condensation heat transfer coefficients of R-600a in coiled wire inserted tubes.

Keywords: Coiled wire; Condensation; Flow regime; Heat transfer; R-600a.

Nomenclature

\dot{m}	mass flow rate (kg/s)
C_p	specific heat (kJ/kg.K)
D	tube diameter(mm)
F	interfacial roughness factor
G	mass flux (kg/m ² s)
H	enthalpy (kJ/kg)
h	heat transfer coefficient (kW/m ² K)
I	current (A)
k	thermal conductivity (W/m.K)
L	length (mm)
Pr	Prandtl number
p	pressure (kPa)
Q	heat transfer rate (W)

q	heat flux (kW/m ²)
Re	Reynolds number
T	temperature (K)
V	voltage (V)
x	vapor quality
P	coil pitch (m)
e	wire diameter (m)

Greek symbols

θ	helix angle of coiled wire (degree)
η	viscosity
ρ	density
γ	efficiency factor

Subscripts

c	condenser
e	equivalent
f	liquid phase
g	vapor phase
h	heater
i	inner
in	inlet
lat	latent
o	outer
ref	refrigerant
sat	saturated
sen	sensible
tot	total
tc	test condenser
w	wall
w	water

1. Introduction

By progressive development of the refrigeration industries, heat exchangers have become an indispensable part of the instruments used for either cooling or heating purposes. In fact, augmenting the performance of the thermal systems employed in refrigeration industries is highly recommended as demands for energy modification has been growing increasingly. Nowadays, various options are provided for the researchers to ameliorate the heat exchangers performance and augment the heat transfer rate such as employing coiled wire inserts [1, 2]. These inserts can be produced easily and cheaply and installed or uninstalled rapidly in order to clean inside tubes. The other merit of these inserts is that the mechanical strength of the primary plain pipe would not be affected adversely [3]. Therefore, spiral coils are advantageous in

industrial applications and need to be studied further. However, it should be mentioned that the use of these devices induces considerable pressure losses.

Two-phase heat transfer enhancement of heat exchangers utilizing coiled wire inserts has already investigated by researchers. To ameliorate the rate of transferred heat through a counter-flow heat exchanger, Agrawal et al. [4] empirically evaluated the condensation of refrigerant R-22 by utilizing coiled wires with various geometrical specifications. It was found that the spring has generally a positive effect on overall performance. According to the reported results, heat transfer coefficient in rough tubes was enhanced 100% in comparison to the primary smooth tube. Yun et al. [5] employed several spiral coils inside a horizontal tube to assess the influence of these instruments on the thermal performance of a test tube for various operational conditions when nitrogen was utilized as the refrigerant. The inserts having varied wire diameters and coil pitches were found to be efficacious on evaporation of the nitrogen. The experiments revealed that the increment of the mass velocity, the decrement of the inserts pitch, and the increment of the wire diameter contribute to further augmentation of the heat transfer coefficient. Another experimental study was done by Akhavan-Behabadi et al. [6] to illuminate the influence of springs on the magnitude of heat transfer coefficient obtained during the evaporation of the high global warming potential (GWP) refrigerant R-134a inside horizontally installed copper tubes. Generally, springs having wire thicknesses between 0.5-1.5 mm and pitches between 5-13 mm improved the rate of transferred heat as compared to the primary test tube without inserts. The highest growth of 98% in coefficient of heat transfer was obtained using a spiral coil with the thickest wire and smallest pitch. Also, the inserts increased the pressure loss. The maximum increase in pressure loss was about 1000% over the smooth pipe. Within another experiment, Akhavan-Behabadi et al. [7] examined the condensation heat transfer enhancement of refrigerant R-134a using springs in horizontal pipes. Similarly, it was reported that the coiled wire having a wire thickness of 1.5 mm and coil pitch of 10 mm enhanced the coefficient of heat transfer about 80% over the plain pipe. Shafaei et al. [8] by performing an experiment on horizontally installed copper pipes assessed the influence of springs on the rate of evaporative heat transfer when varied mass velocities and vapor qualities were applied to the system and R-600a was utilized as the refrigerant. It was observed that higher coefficients of heat transfer were obtained as the mass velocity increased. According to the reported observations, the system reached the highest heat transfer rate by utilizing spiral coils with thicker wires and smaller coil pitches. A visual study was also performed and it was observed that the intermittent and annular flow patterns dominate. Results of another experimental investigation conducted by Jalil and Goudarzi [9] also showed that the use of springs resulted in the significant augmentation of the heat removal. The experiments also showed that by using the coiled wire insert, the friction factor increased in the range of 42-49% over the smooth pipe depending on the Reynolds number. Considering an increment of the performance factor, it was suggested that the use of these devices is advantageous.

The impacts of wire coils usage in single phase flows have been also reported extensively. Hong et al. [10] appraised experimentally the turbulent heat transfer and pressure drop for air in smooth tubes with wire coils having several pitches and width, under the conditions of constant heat flux and variable Reynolds number. The findings illustrated the significant growth of the

By reviewing the literature, it is evident that there is no research conducted previously to evaluate the performance of environment-friendly refrigerant R-600a in spiral coil inserted pipes. In this empirical research, the impacts of circular cross-section spiral coils usage inside the horizontally installed tubes on the heat transfer characteristics of refrigerant R-600a (Isobutane) during forced convection condensation is appraised.

It is noteworthy that refrigerant R-600a is examined in the current study because of the appropriate features which it has. Due to the serious environmental problems such as global warming and ozone layer depletion during recent decades, introducing environment-friendly refrigerants is a crucial point. R-600a has an insignificant GWP. Also, the ozone depleting potential (ODP) of this refrigerant is zero [13, 14]. The thermal conductivity of R-600a is larger than that of some other refrigerants such as R-134a and R-22 which would result in better heat transfer performance. The other advantage of R-600a over the widely-used refrigerant R-134a is having a higher value of the latent heat. As the latent heat of a refrigerant is larger, less mass flow rates would be necessary for producing a specific capacity [15, 16]. Also, Lee et al. [17, 18] reported that the calculated performance of the heat-exchangers working with Isobutane was superior to that obtained by the other refrigerants as R-22. Yu and Teng [19] reported that refrigerant R-600a usage could improve the energy factors of refrigerators. Consequently, R-600a is used in the current research.

2. Experimental setup

In **Fig. 1** a schematic view of the constructed cycle is demonstrated for better understanding of the whole process. The setup includes the condenser, post condenser, heaters, flow meters, reservoir, thermocouples, gear pumps, pressure indicators, sight glasses, and differential pressure drop transducers. Two closed loops constitute the cycle. The first loop is charged with R-600a while the second loop works with the cold water. The cold water is utilized as the coolant so as to eliminate the latent heat of the refrigerant R-600a.

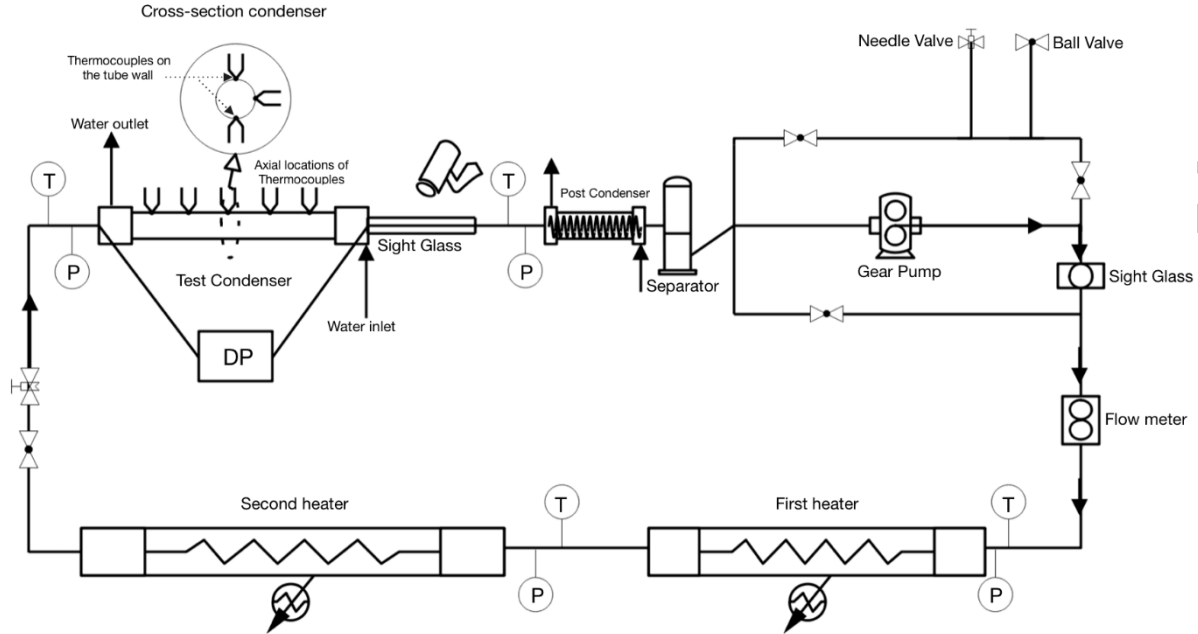


Fig. 1. Schematic configuration of the studied refrigeration cycle

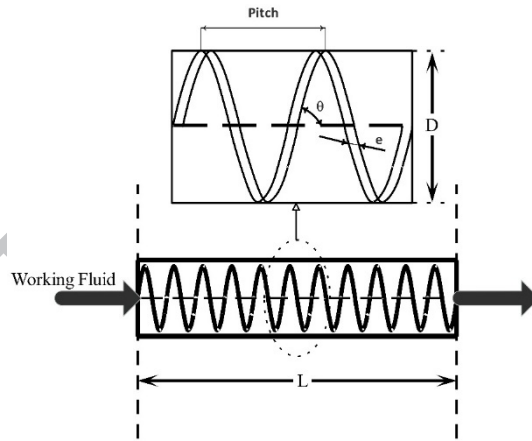


Fig. 2. Physical model of the test tube and the inserts.

Two variable frequency gear pumps were utilized to circulate the refrigerant and cooling water as well as to control their mass velocities. In each loop after the pump, a flow meter having the precision of 0.1% of the full scale was placed for measuring the mass flow rates. Two electrical resistance heaters were located just before the test condenser to obtain the required vapor quality at the test section inlet and insulated carefully to prevent from heat losses. The test condenser was a counter-flow coaxial double-pipe heat exchanger with an inner tube that was made of copper. The length, the internal diameter and the wall thickness of the studied tube were 1000, 8.1, and 0.71 mm, respectively. The refrigerant R-600a flows within the internal tube whereas the

coolant water flows in a counter flow direction within the annulus. To obtain the refrigerant enthalpy at the heaters inlet, a 100 RTD type temperature sensor and a pressure transducer were used. The provided power by heaters was measured using a watt transducer. The tests were conducted using two types of horizontal pipes with the length of 1000 *mm* including the plain tube and tubes with spiral coils (rough tubes). **Fig. 2** illustrates the schematic view of the test tubes and the spiral coils. Geometrical features of the tubes utilized in this experimental research are given in **Table 2**.

Table 2. Specifications of the tubes with coiled wires used in the current study.

Tube set	P (mm)	e (mm)	θ (degree)	Tube inner diameter (D_{in}) (mm)	e^2/PD_{in} (-)
Plain	-	-	-	8.1	-
CW1	10	0.5	67.27	8.1	0.003
CW2	30	1	36.63	8.1	0.0041
CW3	20	1	48.12	8.1	0.0061
CW4	10	1	65.85	8.1	0.0123
CW5	10	1.5	64.25	8.1	0.0185

At five axial locations with a distance of 200 *mm* from each other, T-type thermocouples with the precision of 0.1 °C were attached for measuring the wall outside temperature of the inner tube. In this regard, three thermocouples were attached to the bottom, side, and top of the pipe at each axial position by welding. Furthermore, in order to determine the cooling water temperature at the shell boundaries, two calibrated temperature sensors (RTD PT 100) were placed at these locations. A pressure gauge (EN 837-1 Wika model) was placed just before the condenser so as to capture the pressure loss along the studied test condenser. A reservoir and a counter-flow condenser were placed between the test section and the gear pump to make sure that the working fluid, Isobutane with the purity of 99.55%, is completely subcooled before flowing inside the pump. Also, a sight glass is used after the test tube for capturing the flow patterns. For this purpose, a method proposed by Jassim et al. [20] is used. In this method, an illuminated diffuse white film with black strips is used behind the sight glass for better recognizing the pictures. The photos are taken by a digital camera. Also, the sight glass was made of Pyrex glass with the inner diameter of 8.1 *mm* as the test tube and the length of 200 *mm*. **Table 3** presents the governing operational conditions of this work.

Table 3. The ranges of operational parameters in the current investigation.

Parameter	Type or value	Units
Working fluid	R-600a (Isobutane)	None

Heat flux	15-20	kW/m ²
Refrigerant mass velocity	115-365	kg/m ² s
Saturation temperature	37-40	°C
Average pressure	5.1	bar
Vapor quality	0.03-0.79	None

A suggested method by Schultz and Cole [21] is used to calculate the uncertainties analysis. In order to evaluate the effect of every parameter on the uncertainty, the following equation is used:

$$U_R = \left[\sum_{i=1}^n \left(\frac{\partial R}{\partial V_i} U_{V_i} \right)^2 \right]^{1/2} \quad (1)$$

For a desired variable of R , the uncertainty of its calculated value can be estimated by U_R , owing to the uncertainty summation of n independent variables of V_i . The uncertainty of independent variables is indicated by U_{V_i} in Eq. (1). The calculations indicated that the uncertainties in determining the heat transfer coefficients and vapour quality are below 10% and 6% for the experiments, respectively. In **Table 4** the uncertainty of the test parameters as independent variables (V_i) which were used in uncertainty calculations of Eq. (1) are provided.

Table 4. Uncertainties of the measured quantities.

Parameter	Uncertainty	Units
Thermocouples	0.1%	°C
Power	1% of reading	kW/m ²
Refrigerant mass velocity	0.1% of reading	kg/m ² s
Diameter	0.05	mm
Length	0.5	mm

3. Data reduction

During the experiments, condensation data were deemed to be in steady state conditions while pressures and temperatures of the whole refrigeration cycle were monitored fixed for about 15 minutes. Then the reproducibility of test apparatus was checked by iterating 25% of the test-runs twice. Engineering Equation Solver (EES) was utilized for collecting the thermo-physical and thermodynamic properties of the Isobutane. The heat loss from heaters was assessed by single-phase R-600a flow and therefore the thermal efficiency is as follows:

$$\gamma = \dot{m}_{ref}(H_2 - H_1)/(VI)_h \quad (2)$$

In Eq. (2), \dot{m}_{ref} is the mass flow rate of the refrigerant. V and I represent the electric voltage and current, respectively. Also, H_2 shows the refrigerant enthalpy just before entering the test condenser. It is worth mentioning that H_2 is the enthalpy associated with the sub-cooled refrigerant temperatures and pressures. Furthermore, H_1 is the refrigerant enthalpy, associated with the sub-cooled flow temperatures and pressures, just before entering the heaters. Based on the calculations, thermal efficiency of the evaporator was 0.96.

With considering the thermal efficiency of heaters, the quality at the inlet of the test condenser can be formulated by:

$$x_{in,tc} = (\gamma(VI)_h - C_p \dot{m}_{ref}(T_{sat,h,in} - T_{ref,h,in}))/\dot{m}_{ref}H_{fg,h} \quad (3)$$

Where C_p is the refrigerant specific heat. $T_{sat,h,in}$ shows the refrigerant saturation temperature associated with the heaters mean pressure. $T_{ref,h,in}$ represents the refrigerant temperature just before the heaters. Finally, $H_{fg,h}$ is the refrigerant vaporization enthalpy related to the heaters mean pressure.

Similarly, the heat leakage from the test condenser was taken into account using the equation proposed by Xing et al. [22]:

$$\gamma_c = \frac{\dot{m}_w C_{p,w}(T_{w,out} - T_{w,in})}{\dot{m}_{ref} C_{p,ref}(T_{ref,c,in} - T_{ref,c,out})} \quad (4)$$

In Eq. (4), \dot{m}_w is the water mass flow in cooling water loop. $T_{w,out}$ and $T_{w,in}$ are water outlet and inlet temperatures. $T_{ref,c,in}$ and $T_{ref,c,out}$ show the refrigerant temperature at the condenser inlet and outlet, respectively. Also, $C_{p,w}$ represents the water specific heat. Consequently, the thermal efficiency based on the calculations was 0.95.

In order to define the heat transfer rate of the condenser, an energy balance is performed for the water, which passes through the annulus:

$$\dot{Q}_w = \gamma_c C_{p,w} \dot{m}_w (T_{w,out} - T_{w,in}) \quad (5)$$

In Eq. (5), \dot{Q}_w is the total heat which is transferred from the test condenser. \dot{m}_w is the water mass flow. The water specific heat is represented by $C_{p,w}$ which is taken based on water mean temperature in the annulus. Also, $T_{w,out}$ and $T_{w,in}$ are the water temperatures at the annulus boundaries.

For obtaining the refrigerant qualities at the test section outlet, another energy balance is done for the condenser:

$$x_{out,tc} = x_{in,tc} - \dot{Q}_w/(\dot{m}_{ref} H_{fg,tc}) \quad (6)$$

Where $H_{fg,tc}$ demonstrates the refrigerant vaporization enthalpy associated with the test condenser mean pressures.

For the local vapor quality of the whole test section, the mean of vapor qualities at the test condenser boundaries is considered:

$$x_{tc} = (x_{in,tc} + x_{out,tc})/2 \quad (7)$$

Employing the Said and Azer method [23], the quasi-local convective heat transfer coefficient is obtained by Eq. (8):

$$h_{ref} = \left[\frac{\pi D_i L (T_{sat} - T_{wall})}{\dot{m}_w C_{p,w} (T_{w,out} - T_{w,in})} - \frac{D_i}{2k} \ln \left(\frac{D_o}{D_i} \right) \right]^{-1} \quad (8)$$

Where L is the test pipe length. D_i demonstrates the pipe internal diameter. D_o is the pipe outer diameter. k shows the copper pipe thermal conductivity. T_{sat} is the refrigerant saturation temperature associated with the test section pressures. Finally, T_{wall} represents the test pipe temperature which is determined by calculating the mean of captured temperatures by attached thermocouples on the test pipe.

4. Results and discussion

The experimental data are obtained by using R-600a as the refrigerant at mass velocities of 115, 154, 268 and 365 kg/m^2s and vapor qualities within the range of 0.03-0.79. The data of heat transfer coefficients are collected for both the plain tube and coiled wire inserted tubes.

4.1. Heat transfer in smooth and rough pipes

The comparison between the obtained experimental results for the plain pipe and the data predicted by Thome et al. [24] and Cavallini et al. [25] correlations is illustrated in Fig. . These correlations are usable for hydrocarbon refrigerants. The model proposed by Thome et al. [24] could predict the condensation heat transfer coefficients for a wide range of flow regimes. Also, it is noteworthy that this model is evaluated for wide ranges of mass fluxes, vapor qualities and tube internal diameters, which satisfies the operating parameters of the current research. As it was expected, more than 88% of the obtained condensation heat transfer coefficients are inside $\pm 30\%$ error window. The other model proposed by Cavallini et al. [25], was not as precise as the first model in predicting the condensation heat transfer coefficients. However, this model was capable of predicting the 52% of experimental results within $\pm 30\%$ error range. It is worth noting that annular, intermittent, and stratified-wavy regimes were experienced in the present research. Fig. 4 represents the two-phase flow patterns observed in this experiment. At the high vapor quality of 0.78, the annular flow pattern is observed. The reason is that when the vapor quality is large, the liquid volume will be small. Consequently, the shear stress between the liquid and vapor phases dominates the gravity force and the liquid extends around the periphery of the tube and the vapor phase flows in the core. However, as the vapor quality decreases the liquid volume increases which results in augmenting the effect of gravity force. So, the gravity force dominates

the shear stress between two phases and the large volume of the liquid flows at the bottom of the tube.

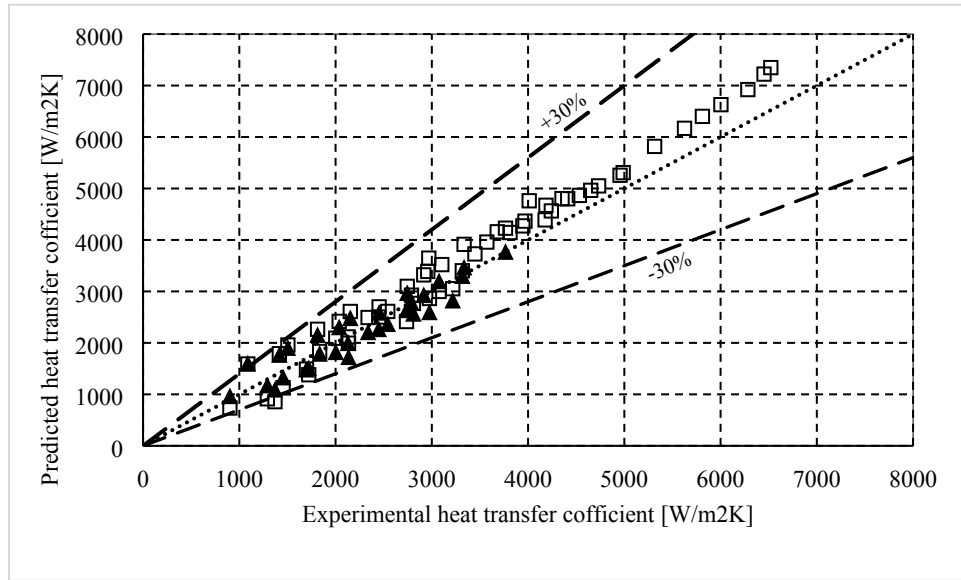
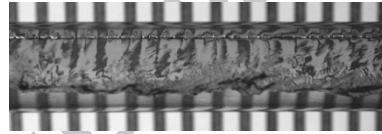
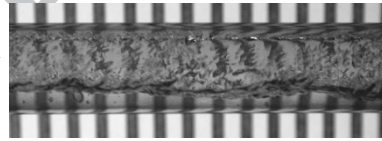


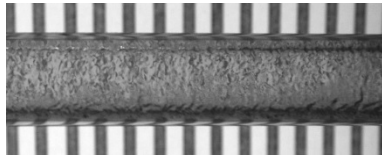
Fig. 3. Comparison between the heat transfer coefficients obtained by experiments and correlations.



Stratified-wavy ($G=154 \text{ kg/m}^2\text{s}$, $x=0.242$)



Intermittent ($G=154 \text{ kg/m}^2\text{s}$, $x=0.32$)



Annular ($G=154 \text{ kg/m}^2\text{s}$, $x=0.78$)

Fig. 4. The observed flow patterns in this study.

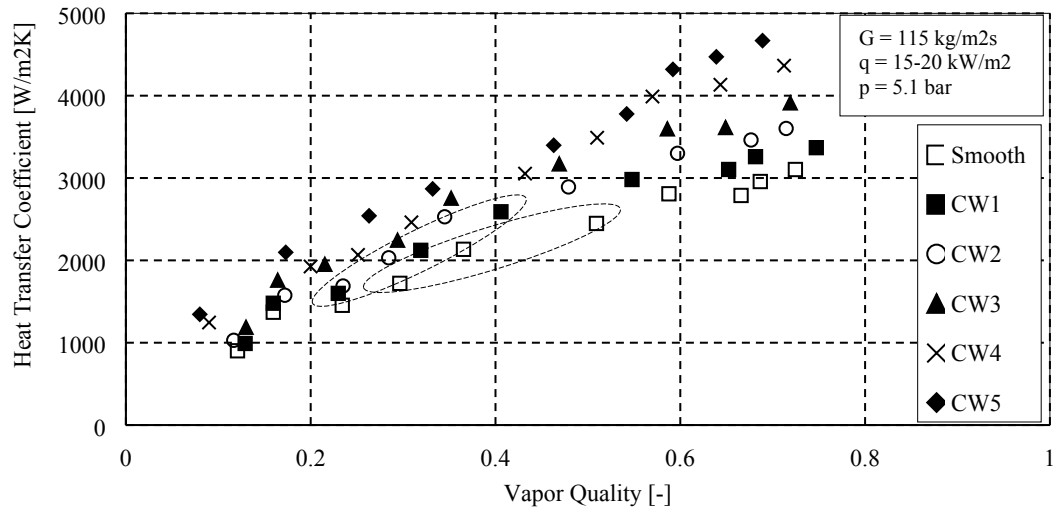


Fig. 5. The trend of heat transfer coefficients changes with changes of vapor quality for mass velocity of 115 kg/m²s.

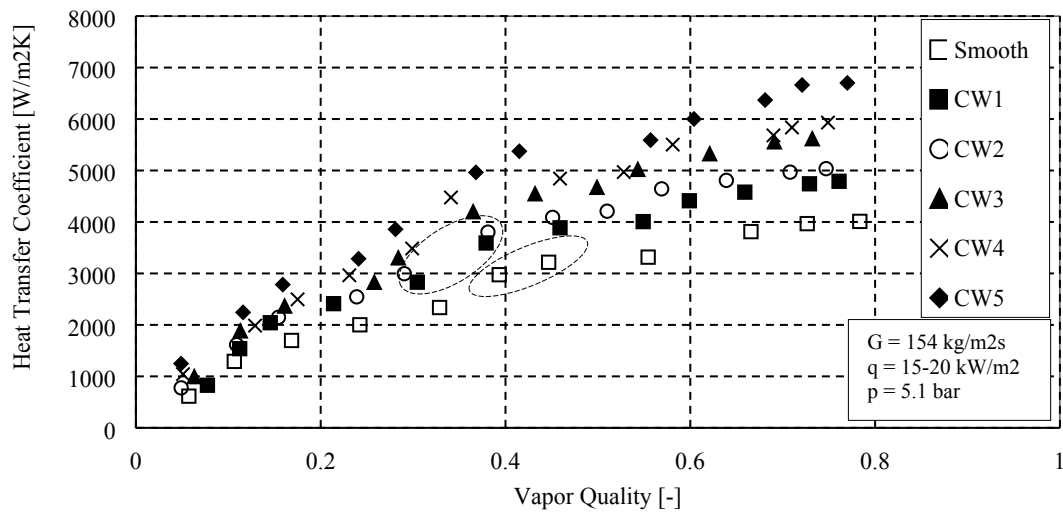


Fig. 6. The trend of heat transfer coefficients changes with changes of vapor quality for mass velocity of 154 kg/m²s.

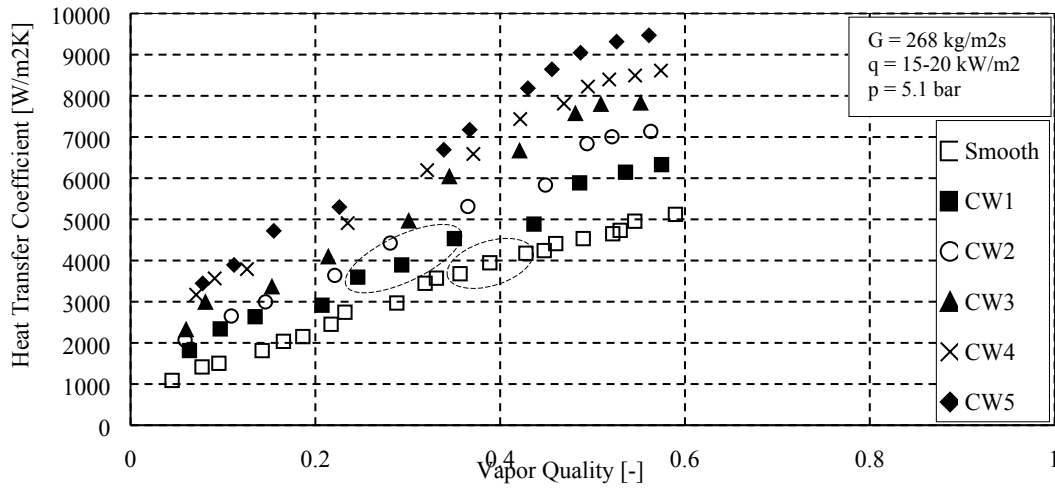


Fig. 7. The trend of heat transfer coefficients changes with changes of vapor quality for mass velocity of 268 kg/m²s.

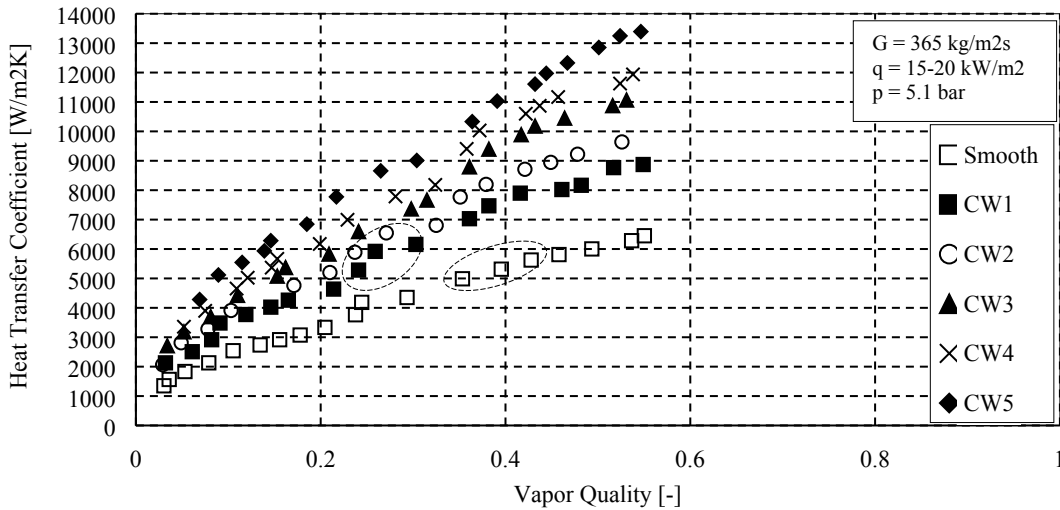


Fig. 8. The trend of heat transfer coefficients changes with changes of vapor quality for mass velocity of 365 kg/m²s.

According to Figs. 5-8, it could be seen that for all mass velocities, the increase of the vapor quality contributes to the enhancement of heat transfer coefficients for the smooth tube and rough tubes regardless of the insert types. This behavior is firstly due to the fact that the higher vapor quality leads to the establishment of a thinner liquid film on the pipe internal wall. Consequently, the thermal resistance reduces. The other reason is that the growth of vapor quality results in the reduction of the mean flow density by augmenting void fraction. Consequently, the fluid flow would accelerate increasing convection transport from the tube wall. It is worthy of attention that the magnitude heat transfer growth is related to the flow regimes and patterns, and it increases with the increase of vapor quality. Similar results were

reported in the experimental study of Son and Oh [26] who evaluated the condensation of CO₂ in horizontal smooth and rough pipes. The annular flow pattern dominates when the vapor quality is high. So, for the annular flow regime higher heat transfer coefficients are obtained. For intermittent and stratified-wavy flow patterns where the vapor quality is lower, smaller heat transfer coefficients are obtained. According to the results, the maximum heat transfer increases (e.g. CW1 from 11 to 41% for all mass fluxes) occurred at the areas where the vapor qualities were high exactly before when the heat transfer modes were changing from annular to intermittent during the condensation. It is noteworthy that the transitions from annular to intermittent for “CW1”, as a typical example compared to the smooth tube for entire mass velocities, were observed through the sight glass after the test condenser and are marked with ovals on the **Figs. 5-8**. In general, the magnitude and the rate of heat transfer curves with vapor quality for various coiled wire pipes are greater than those for the smooth pipe in all mass fluxes, and they all follow some particular trends. This is because of the unlike geometrical aspects of every different coiled wire inserts which will be discussed in depth in the following.

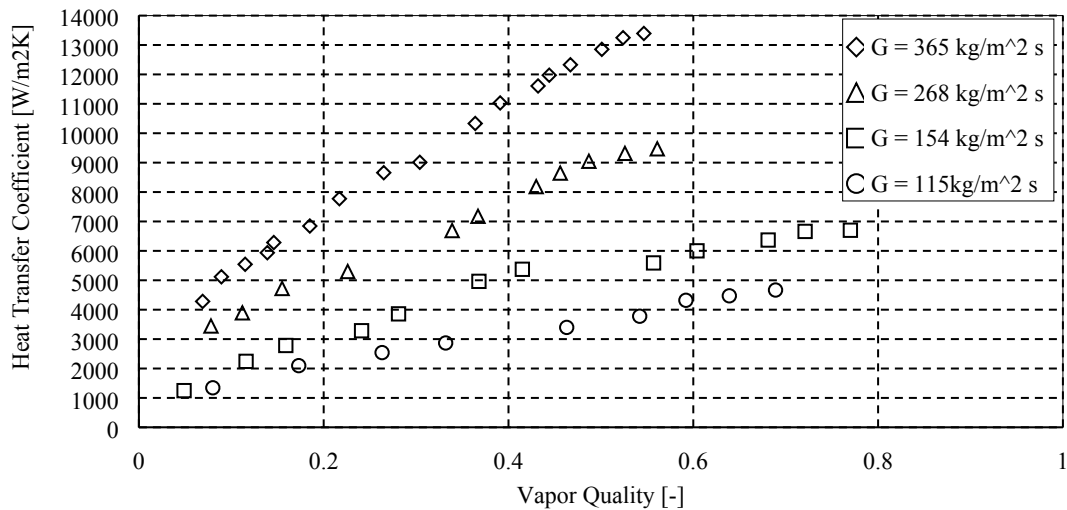


Fig. 9. The trend of heat transfer coefficients changes with changes of vapor quality for various mass velocities for the case of “CW5”.

The trend of the heat transfer coefficients with the vapor qualities change, for the case of “CW5” as the best performing insert, is showcased in **Fig.** by considering the R-600a mass velocity as a variable parameter. As it observable, the coefficients of heat transfer grow as the R-600a mass flux increases for all values of the vapor qualities. Indeed, the increment of the refrigerant mass flux contributes to a higher turbulence intensity in both phases of liquid and vapor. Thus, the transferred heat and the condensation have enhanced.

By considering **Figs. 5-8** it is observed that for any given vapor quality and for all inserts regardless of their geometrical features, the heat transfer augments as the mass velocity increases. It is worth mentioning that by increasing the mass velocity the swirl flow becomes stronger helping the heat transfer. However, decreasing the coil pitch, increasing the wire

diameter along with increasing the mass velocity further intensify the swirl flow resulting in significant enhancement of the heat transfer.

The impacts of coiled wires on the heat transfer coefficients are demonstrated in **Figs. 5-8**. As it is evident, all of the springs have a constructive effect on the heat transfer rate. In this regard, “CW5”, the coiled wire with the wire thickness of 1.5 mm and the coil pitch of 10 mm, performs better compared to other inserts by increasing the heat transfer coefficient by about 55-107% above the plain pipe. The heat transfer improvement depends complexly on the physical characteristics of the coiled wires as well as vapor quality and mass velocity. Coiled wires influences on the coefficients of heat transfer are much considerable at regions with higher vapor qualities. Generally, different trends of heat transfer enhancement at areas with higher and lower vapor qualities are seen. From **Figs. 5-8**, it is observable that the maximum coefficient of heat transfer is obtained at regions with the higher values of the vapor qualities. At these regions, the spring is not completely covered with the liquid refrigerant on the pipe wall and a part of that would be out of liquid phase. Therefore, a portion of spring will be in the vapor core. In this situation, not only does the coiled wire interrupt the laminar layer, but also substantially promotes the turbulence of liquid film resulting in significant enhancement of the transferred heat.

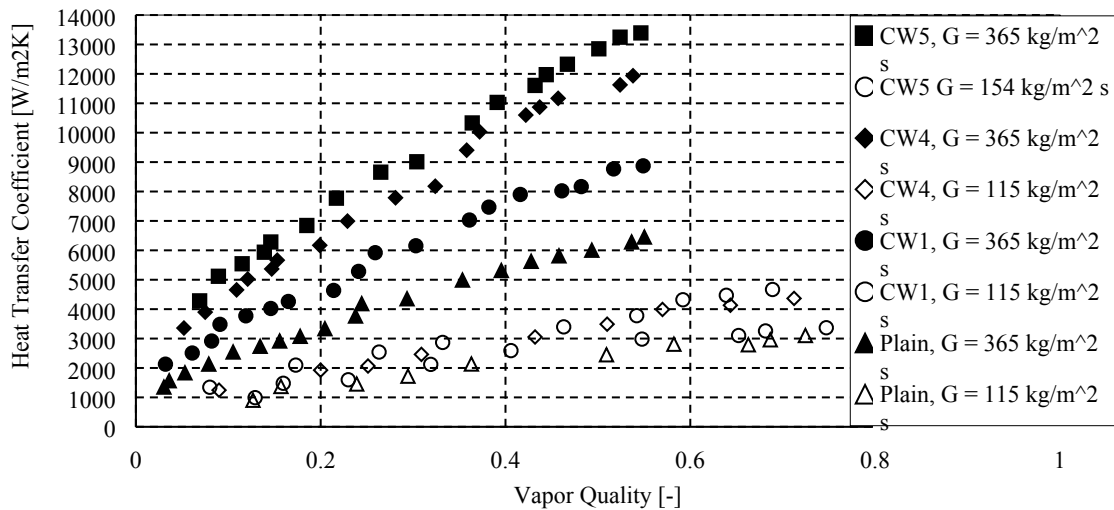


Fig. 10. The trend of heat transfer coefficient changes with changes of vapor quality for various wire thicknesses of 0.5 (CW1), 1 (CW4) and 1.5 mm (CW5).

Fig. 10, illustrates the effects of springs having constant pitch of 10 mm and various wire thicknesses of 0.5 (CW1), 1 (CW4) and 1.5 mm (CW5) for mass fluxes of 115 and 365 kg/m²s. Results indicated a direct relation between the enhancement of transferred heat and the wire thickness (e). Similar observations were reported by Yun et al. [5] during the boiling of nitrogen inside tubes with springs. The increment of the wire thickness contributes to the heat transfer coefficient augmentation. For the mass fluxes of 115 and 365 kg/m²s, the maximum growth of heat transfer coefficient for the tube set “CW5” are 55% and 107%, respectively, while these values become 11% and 41% for the tube set “CW1”. Actually, the increase of the wire thickness

contributes to the further interruption of the boundary layer development on the tube inner wall during the flow condensation. As the laminar sub-layer of the liquid film that is formed on the tube inner wall becomes more turbulent and thinner, the thermal resistance will be lower which leads to the higher values of the heat transfer coefficients.

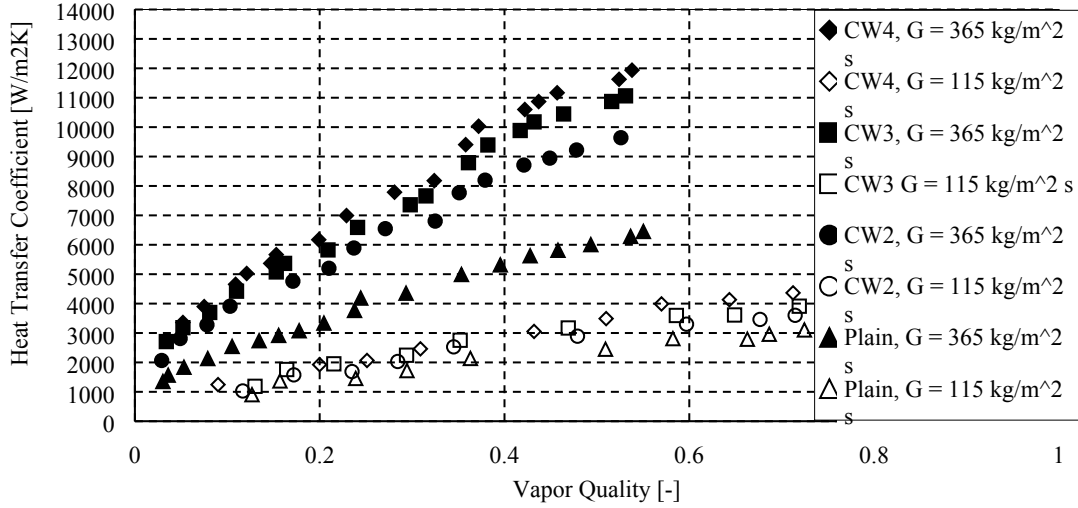


Fig. 11. The trend of heat transfer coefficient changes with changes of vapor quality for various coil pitches of 10 (CW4), 20 (CW3) and 30 mm (CW2).

Fig. demonstrates the influences of coiled wires which own constant wire thicknesses of 1 mm and various pitches of 10 (CW4), 20 (CW3) and 30 mm (CW2) on the heat transfer coefficient when mass velocities are 115 and 365 kg/m²s. Results show the growth of the heat transfer coefficients with decrement of the springs pitch.

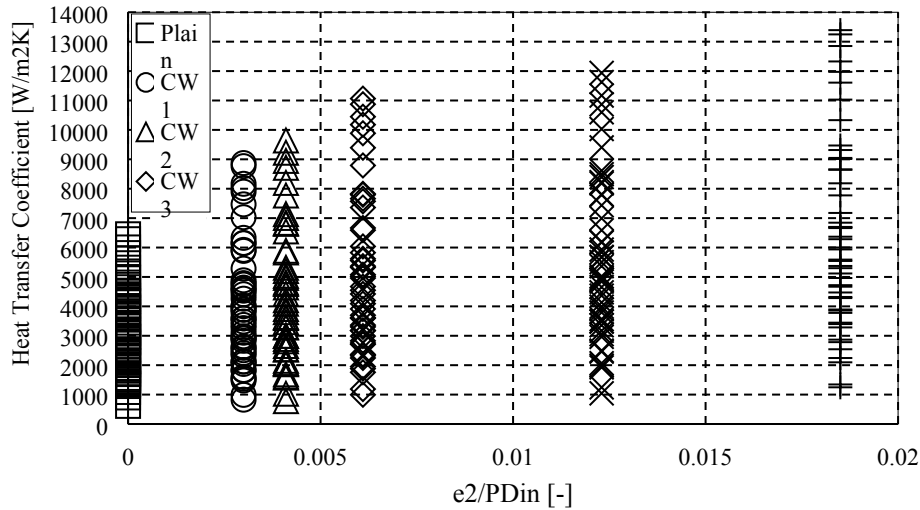


Fig. 12. The boundary of heat transfer growth with coiled wires index under different mass fluxes and vapor qualities studied in the current research.

In order to delve deeply into the effects of coiled wires geometry, a dimensionless parameter i.e. coiled wire index, e^2/PD_{in} , is defined. This parameter is dependent to the ratio of the wire thickness to coil pitch, e/P , and the wire thickness to the tube inner diameter, e/D_{in} . In this regard, **Fig. 4** is plotted to illuminate the effects of coiled wires geometry on the coefficient of heat transfer. It is observed that there is a direct relation between the spring index and the heat transfer increase. The higher the spiral coil index, the greater the boundary of heat transfer augmentation. By considering **Fig. 4**, it can be concluded that the wire diameter, e , has a prominent influence on the heat transfer augmentation, as the index is proportional to the square of the wire diameter. On the other hand, this parameter is inversely proportional to the coil pitch. It means the heat transfer coefficient enhances as the coil pitch reduces. According to **Fig. 4**, the coiled wire “CW5” with the greatest wire diameter and the smallest pitch is superior to other coiled wires.

It is worth mentioning that not only does the installation of spiral coils affect the heat transfer coefficient, but also influences the flow regime during the condensation. According to the visual observations through sight glass, at high vapor qualities, the annular flow pattern was observed for the smooth tube and rough tubes. Overall, the annular flow pattern depends on flow conditions in which the mass velocities and vapor qualities are higher, as under such circumstances the shear stress between two phases dominates the gravity force. Consequently, the liquid would extend around the periphery while the vapor phase stays in the core. However, the flow pattern transition from annular to intermittent occurs as the vapor quality decreases gradually. This flow pattern transition for the smooth tube happens at the vapor quality of about 0.4 for the mass flux of $115 \text{ kg/m}^2\text{s}$ as it is determined in **Fig. 5** by an ellipse. By the insertion of coiled wires this transition is observed at lower vapor qualities. For instance, for the mass velocity of $115 \text{ kg/m}^2\text{s}$, the transition of annular to intermittent is shown at the vapor quality of around 0.3 by using coiled wires. Indeed, the annular flow pattern is observable at lower vapor qualities when coiled wires are used. In coiled wire inserted tubes, a decrement in liquid film velocity is expected because of the coiled wires presence at the pipe circumference. As a consequence of this, i.e. decrement of the liquid film velocity, and increment of the surface tension impacts, the liquid film thickness in coiled wire inserted tubes would be more than that of the smooth tube prior to flow pattern alteration from annular to intermittent. Therefore, the annular flow regime could be observed even at lower vapor qualities. Similar trends are observed for different mass velocities.

4.2. A new correlation for predicting the rough pipes heat transfer coefficients

The literature study shows that there is no correlation in order to predict the heat transfer coefficients during forced convective condensation of R-600a within spiral coil inserted pipes. To develop such a relation, the suggested correlation by Thome et al. [24] for predicting the heat transfer coefficients (h_s) in plain pipes is used as the primary relation. Then the following functional relationship is implemented:

$$h_r = c_1 h_s^{c_2} \left(\frac{e^2}{PD_{in}} \right)^{c_3} \quad (9)$$

Where e , P , and D_{in} represent the wire diameter, coil pitch, and the tube inner diameter, respectively. Also, h_r is the heat transfer coefficient in the rough tubes.

Based on the current empirical results and by utilizing the least squares regression analysis, the new correlation is suggested as follows:

$$h_r = 1.216h_s^{1.144} \left(\frac{e^2}{PD_{in}} \right)^{0.187} \quad (10)$$

Fig. 13 shows that this correlation is capable of predicting the most of the heat transfer coefficients in an error range of $\pm 20\%$.

To determine the average deviation (AD) and average absolute deviation (AAD) of the results obtained by Eq. (10) from the experimental results, the following relations are used:

$$AD = \frac{1}{N} \left(\sum_{i=1}^N \frac{h_{calculated} - h_{experimental}}{h_{experimental}} \right) \times 100 \quad (11)$$

$$AAD = \frac{1}{N} \left(\sum_{i=1}^N \frac{|h_{calculated} - h_{experimental}|}{h_{experimental}} \right) \times 100 \quad (12)$$

Results illuminated that the average deviation and average absolute deviation of the predicted heat transfer coefficients from the experimental data are -0.45% and 6.23%, respectively.

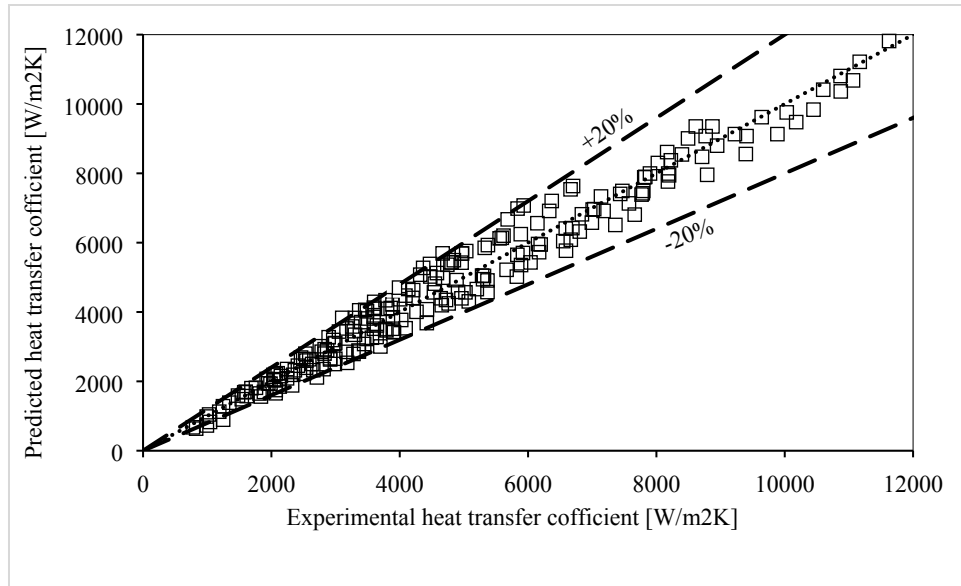


Fig. 13. The comparison of the heat transfer coefficients obtained by the developed correlation and the experiments.

5. Conclusions

The current empirical work is conducted to reveal the influences of spring inserts on the condensation heat transfer characteristics of refrigerant R-600a. In this regard, five different coiled wires with different coil pitches of 10, 20 and 30 mm and various wire thicknesses of 0.5, 1, and 1.5 mm were employed. During the experiments, the vapor quality varied between 0.03-0.79. Also, four mass fluxes of 115, 154, 268, and 365 kg/m²s were considered. The conclusions drawn from this experiment are as follows:

- The increase of the mass velocity and vapor quality resulted in enhancement of the condensation heat transfer for both plain tube and rough tubes.
- The coiled wires contributed to augmentation of the condensation heat transfer coefficient. The maximum heat transfer coefficient enhancement of 107% over the plain tube was observed by using an insert with the greatest coiled wire index of 0.0185.
- The heat transfer coefficient enhanced with increment of the wire thickness and decrement of the coil pitch for any given mass velocity and vapor quality. However, the magnitude of this increment was strongly dependent to the operating conditions of the system.
- The insertion of coiled wires affected the flow regime transition from annular to intermittent. By insertion of coiled wires, annular flow pattern was observed at lower vapor qualities.

According to the experimental results observed in the current study, although inserting the coiled wires significantly contributes to the heat transfer growth, increases in the pressure drops are also observed. The effects of the coiled wires usage on the pressure drop, and the system overall

performance assessment will be discussed in future studies to reveal the overall effectiveness of the coiled wires.

Conflict of Interests

‘Declarations of interest: none’

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